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A Study of Automobile Clutches

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A STUDY OF AUTOMOBILE CLUTCHES

BY

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THESIS

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

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IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF Bachelor of Science in Mechanical Engineering

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A STUDY OF AUTOMOBILE CLUTCHES.

Introduction

One of the most important parts of the automobile is the clutch since it is the connecting link between the power plant and the transmission. Its object is to bring the car up to speed, gradually and without jerking; then transmit the required power without slippage.

It is the object of this thesis to investigate the present tendencies of design and to arrive at some results which will show what the designers and manufacturers of some of our popular cars are doing with reference to the coefficient of friction; and the energy transmitted per square inch of friction area, and to establish if possible the relation existing between these values and the horse power.

CHAPTER I

Classification of Clutches.

1. Axial Clutches. - An axial clutch is defined as one in which the force or pressure is applied parallel to the axis of rotation. Under this heading come cone clutches, disc clutches, and the combined cone and disc clutch.

The cone clutches may again be divided into three sub-classes, the direct cone, inverted cone, and the double cone clutch.

Under the disc clutches there are the multiple disc, and the single disc or plate clutch.

Another type belonging to this class is the combined cone and disc clutch known as the Hele-Shaw clutch.

2. Rim Clutches. - A rim clutch may be defined as one in which the pressure is applied at right angles to the axis of rotation.

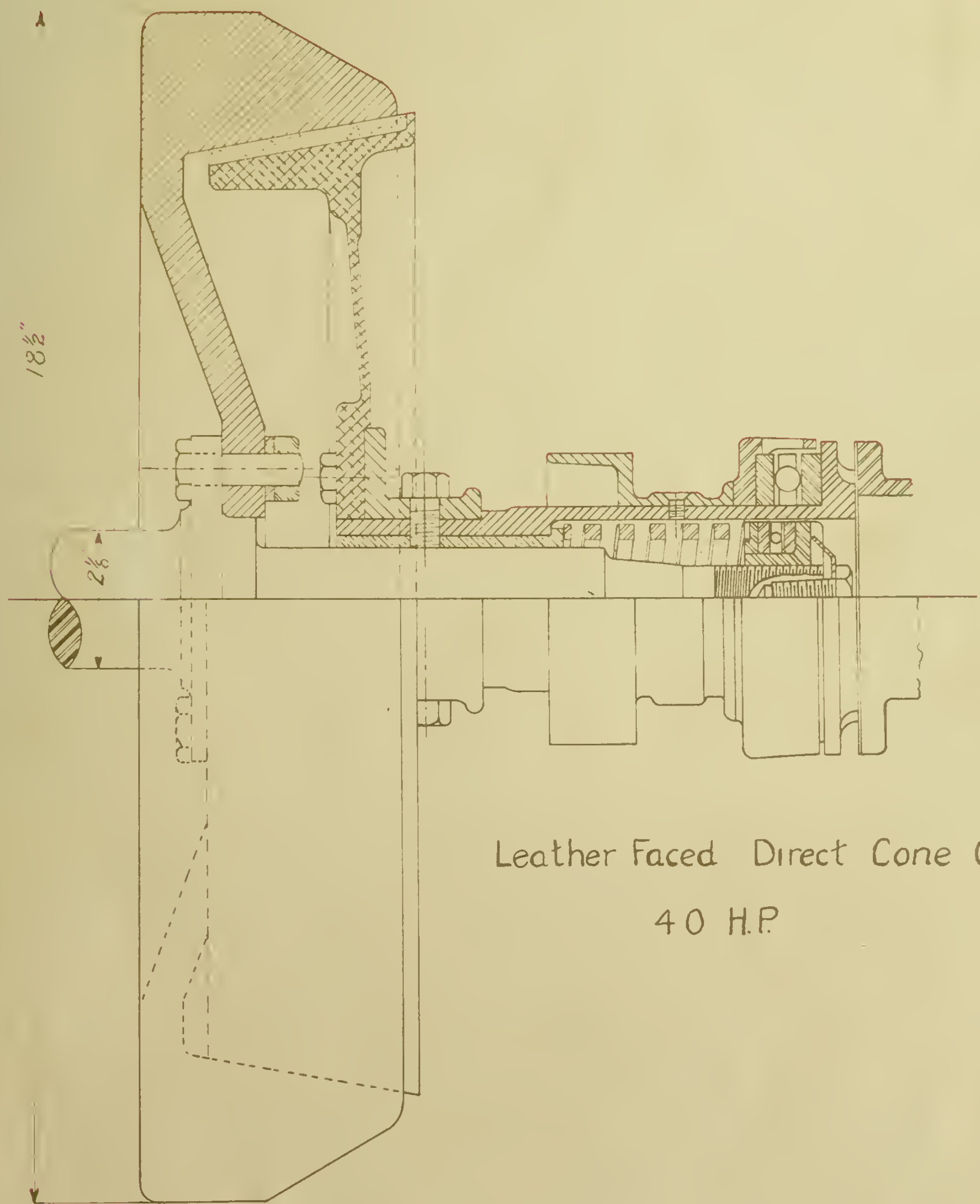
In this class is the expanding band clutch which may have either a single or double band expanding into the rim of the flywheel or drum.

Another type is the external contracting band clutch in which the outside band contracts and thus takes up the load.

CHAPTER II.

Cone Clutches

3. Direct Cone Clutch. - On page 4 is shown a reproduction of a drawing of a direct cone clutch. The flywheel is bored out on the inside rim to receive the clutch spider, which is generally faced with leather, or leather with cork inserts. The cone angle varies with different manufacturers and in the clutches investigated it ran from 10° to 13°. The clutch spider is made as light as possible to reduce the inertia forces and for this purpose aluminum is generally used. The spider is bolted to a sliding sleeve which is keyed to the transmission shaft; and is free to move axially. A spring, provided with a ball thrust bearing holds the clutch in engagement. To release the clutch, pressure is applied to the pedal, which is connected by suitable links to the sliding sleeve, which in turn pulls the spider away from the flywheel.



Leather Faced Direct Cone Clutch

40 H.P.

PLATE I

The clutch is operated in the following manner. When the car is started, the engine is revolving at a high speed. The foot pedal is pressed down, thus throwing the clutch out. The low gear is then thrown in, and the foot pedal slowly released. This gradually forces the spider into the flywheel, due to the spring pressure. If the foot pedal is released suddenly the entire load is thrown on at once and the car starts with a jerk, causing serious strains on the engine, gears and tires. Several devices have been brought out to make the starting more easy. Springs are placed under the leather facing which raises it at several points, and these places come in contact first. The raised surfaces cannot carry the full load but are sufficient to start the motion of the car. As the clutch is completely thrown in the springs are forced down and the full friction surface comes into contact. Some makers use flat springs and others use helical springs, but the purpose is the same whatever the details of construction may be.

4. Force Analysis. - In figure 1 are shown the different forces acting on the direct cone clutch.

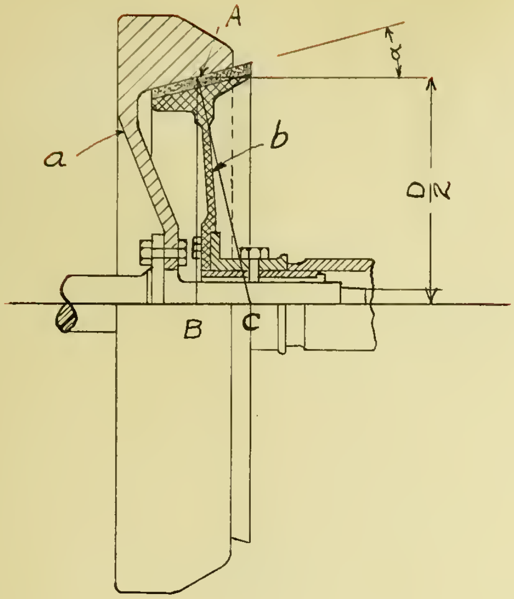


FIGURE 1

Let M = twisting moment to be transmitted.

P = axial force pressing the cones together.

R = normal reaction on each side of cone.

T = total tangential resistance between cones.

α = the cone angle as shown in Fig. 1.

μ = coefficient of friction.

Consider B C = 1/2 P as acting at the point A and the other half of P as acting at a point diametrically opposite to A; then

$$P = 2\,R \sin\alpha \qquad (1)$$

Now from our theory of friction, we know that

$$T = 2\mu R = \frac{2\,M}{D} \qquad (2)$$

Combining equations (2) and (1) we get

$$P = \frac{2\,M \sin\alpha}{\mu D}$$

From the preceding equation it follows that

$$\mu = \frac{T \sin\alpha}{P} \qquad (3)$$

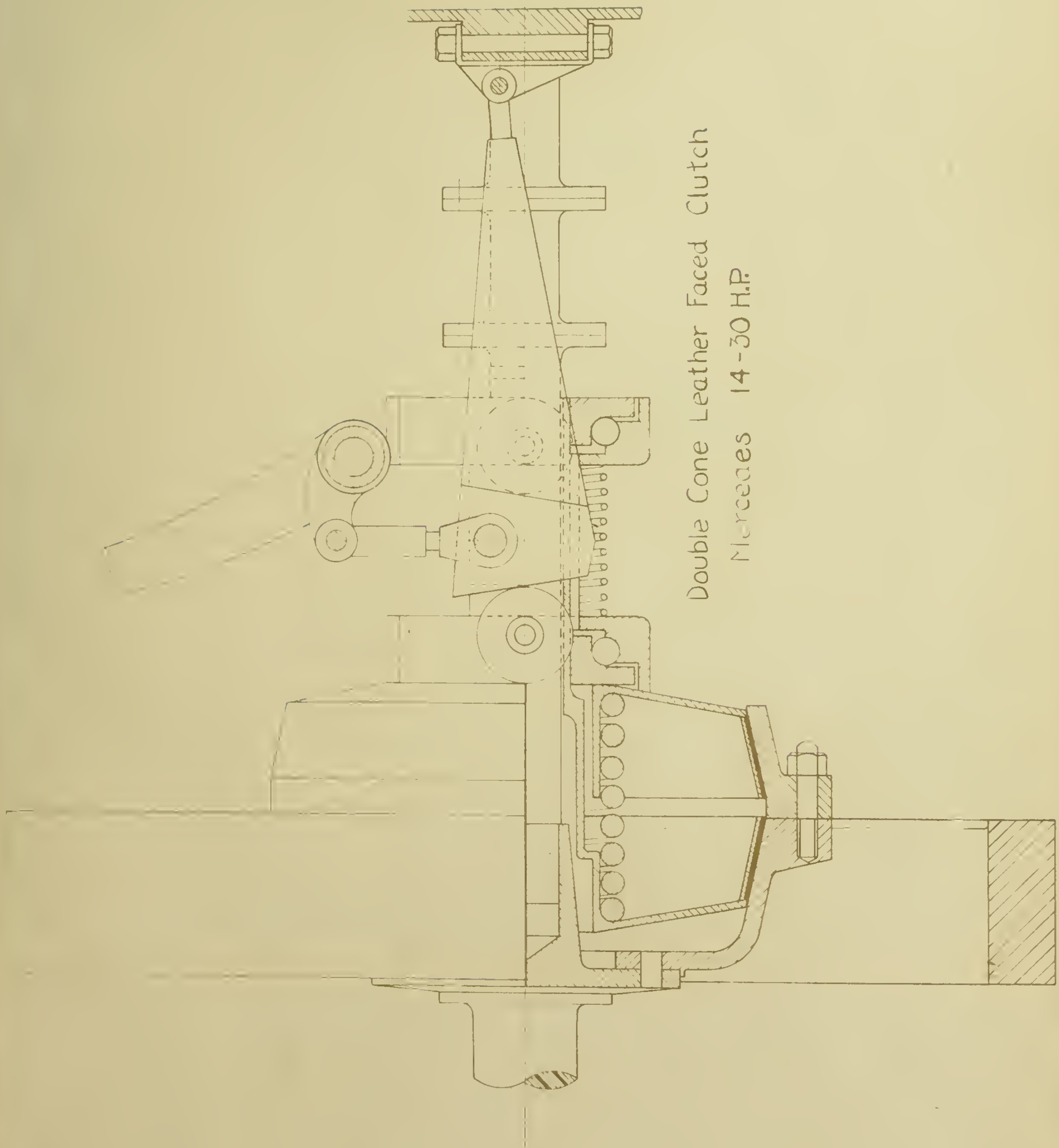


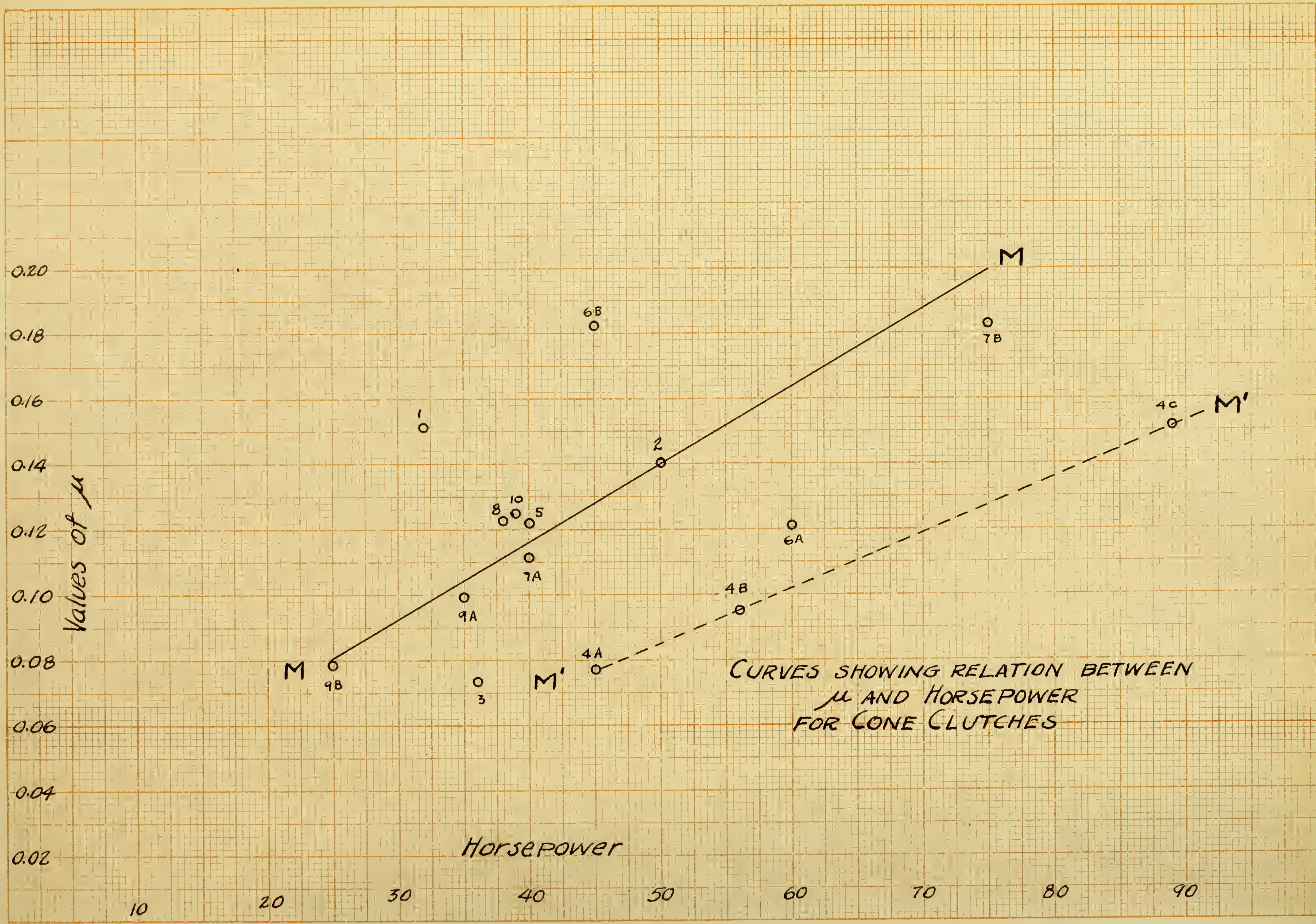
PLATE II

The numerical value of T may be readily calculated from the horsepower transmitted at a given speed. Furthermore P is known since it represents the compression in the clutch spring. Hence knowing the angle α, the value of the coefficient of friction μ may be determined from (3) above.

5. Double Cone Clutches. - Plate II page 7 shows a reproduction of a drawing of a double cone clutch. The flywheel has a drum bored out to receive one of the driving cones. Another drum is bolted to the flywheel and is bored out to receive the other inner cone. The cones are connected to two sleeves which slide axially - one inside the other. The spring on the shaft tends to push the cones apart and forces them against their respective drums. The clutch is thrown out by means of the cams and rollers shown. The cams are bevèled on both sides so the cones are moved towards each other and away from the drums.

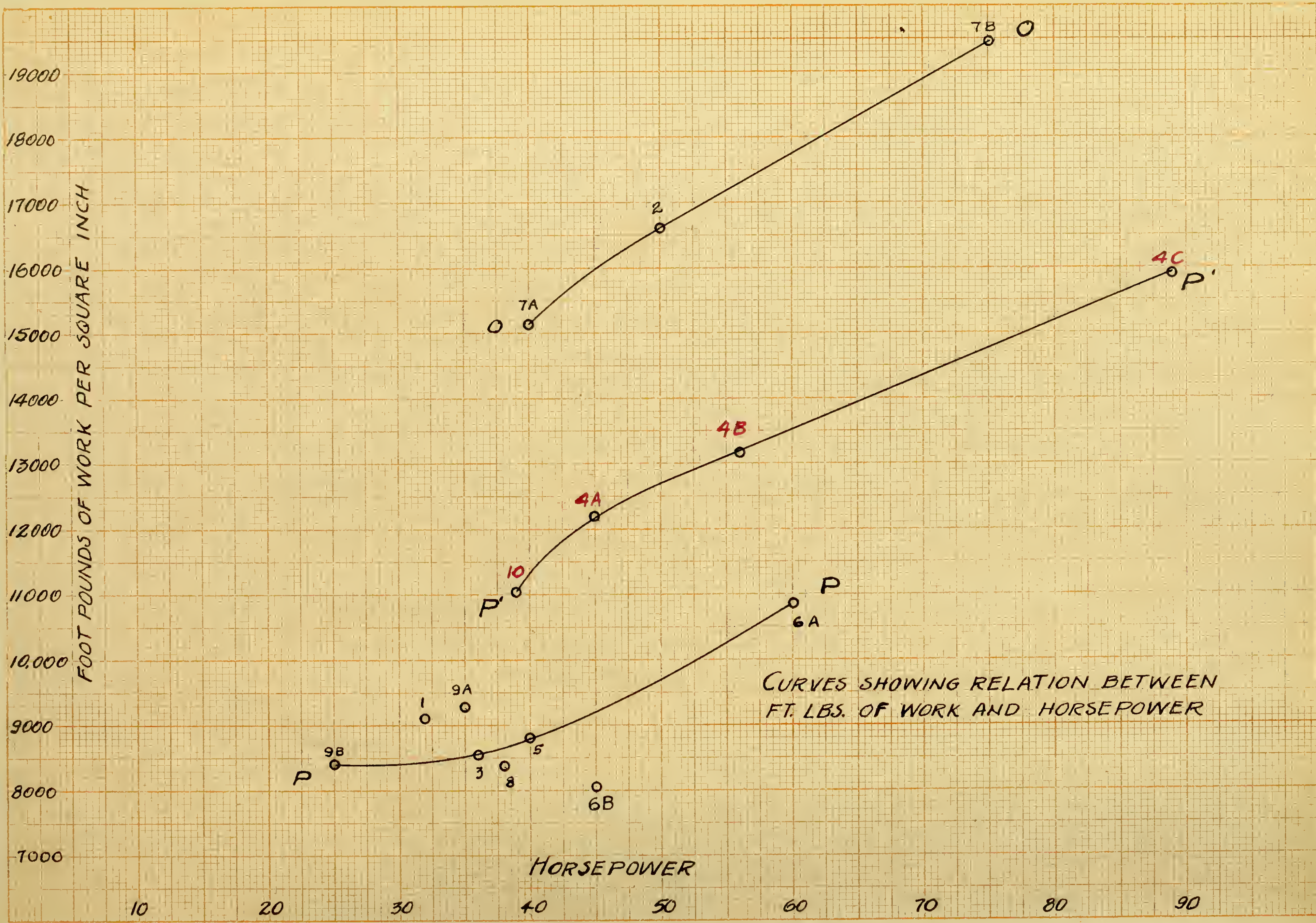
No data relating to horsepower transmitted, speed and spring pressure was received for the double cone clutch hence no investigation was possible. It is used very little and none of the first class cars have it. The principal advantage of this type lies in the fact that it offers more area for contact and gives a capacity for higher horsepower than the ordinary direct cone clutch.

6. Inverted Cone Clutches. - The inverted cone clutch acts away from the flywheel instead of towards it as the direct cone clutch. The drum is bolted to the flywheel and contains the inner or driving cone. The chief advantage of this type over the direct cone is that it brings the gear box nearer the engine by having the spring inside the cone. This type of clutch is not


CURVES SHOWING RELATION BETWEEN
μ AND HORSEPOWER
FOR CONE CLUTCHES
Values of μ
Horsepower
0.20
0.18
0.16
0.14
0.12
0.10
0.08
0.06
0.04
0.02
10
20
30
40
50
60
70
80
90
M
M
M'
M'
1
2
3
4A
4B
4C
5
6A
6B
7A
7B
8
9A
9B
10
U. OF I. S. S. FORM 3

very common and no information can be obtained regarding it.

7. Discussion and Conclusions. - On page 9 curve M M represents the relation between the horsepower and the values of the coefficient of friction for plain leather facing. Curve M' M' shows a curve for the leather facing with cork inserts. Point 10 also represents a leather cork facing but lies very much above thecurve through the other three points which are values calculated from the data furnished by one manufacturer. The makers of cork insert clutches claim a much higher coefficient of friction than the ordinary leather facing but the curves certainly do not bear out this claim. On the contrary all values on the curve M'M' are lower than points on M M for the same horsepower. The points 1, 6B 6A, and 3 are quite a little off of the curve M M and show that the values are either taken close to the limit of general practice or are more than safe. The curve M M is an approximation, but values selected on it would give a satisfactory clutch since it represents average values which are used by first class cars. It is evident that in the cone clutch the value for μ increases considerably with the horsepower. The points above the curve seem to indicate that a higher value for the coefficient of friction could be safely used in the design of clutches of this general type, since all clutches about which information was available are operating successfully in every day use. On page 11, the curves shown represent the relation existing between the horse power and the energy transmitted per square inch of friction area. Curves O O and P P represent values calculated for leather facing and P' P' for the leather with cork inserts. No information could be obtained as to what percentage of the total area



CURVES SHOWING RELATION BETWEEN
FT. LBS. OF WORK AND HORSEPOWER
FOOT POUNDS OF WORK PER SQUARE INCH
HORSEPOWER
19000
18000
17000
16000
15000
14000
13000
12000
11000
10,000
9000
8000
7000
10
20
30
40
50
60
70
80
90
O
7B
2
7A
O
P'
4C
4B
4A
10
P'
P
6A
9B
P
1
9A
3
8
5
6B

in contact was cork, so in calculating these constants, the total area of the conical surface was taken. As a matter of fact the cork area runs from 20 to 30% of the total area. If only the area of the cork inserts were taken the curve would be much higher than it is shown. The curve P'P' being considerably higher than P P shows that the cork will take more work per square inch without slipping. The curve PP may be assumed to represent the average values used in practice at the present time. There is a distinct tendency for the curve to rise above 40 horsepower and this illustrates very well the limitations of the cone clutch with plain leather facing. Around 80 or 90 horsepower the work per square inch is excessive and requires that a high coefficient of friction be used. Curve M M shows that at 75 horsepower the coefficient of friction is about 0.20 and above this it is not safe to work according to the data received. The curve O O represents rather extreme values and beyond this trouble would probably result from heating up of the clutch and slippage, although the three clutches using these high values have anaverage coefficient of friction, and apparently give good service in every day use.

In conclusion it might be said that the cone clutch with leather or leather-cork facing can be safely used up to about 90 horsepower and the coefficient of friction can be taken up to 0.20 without trouble and with a liberal allowance for safety. The energy absorbed per square inch of friction surface for leather should not run much over 11000 pounds for horsepower up to 70. Cork inserts will take more work per square inch running up to 16000 at 90 horsepower.

Pages 14 and 15 show tables giving dimensions of the conical clutches investigated and the resulting values for the coefficient of friction and the work per square inch of friction area.

TABLE OF RESULTS

IDENTIFICATION NUMBER		H.P. TRANSMITTED	COMPRESSION IN SPRING	R P M	ANGLE α	FORMULA	VALUE OF μ	MATERIAL	FT. lbs WORK PER SQ INCH	REMARKS
1		32	200	1200	$12\frac{1}{2}°$	$\mu = \frac{T \sin\alpha}{P}$	.1515	L	9100	
2		50	350-400	1200	13°		.1404	L	16650	
3		36	558-600	800	10°		.0736	L	8570	
4	A	45	640	1000	12°		.0768	L-C	12200	
	B	56	640	1000	12°		.0956	L-C	13120	
	C	89	640	1000	12°		.1517	L-C	15900	
5		40	300	1000	10°		.1218	L	8800	
6	A	60	450	1000	10°		.1215	L	10870	
	B	45	225	1000	10°		.1825	L	8040	
7	A	40	350	1000	11°		.1144	L	15120	
	B	75	400	1000	11°		.1822	L	19420	
8		38	350	850	$10\frac{1}{2}°$		.1223	L	8390	
9	A	35	400	1000	$12\frac{1}{2}°$		.0997	L	9280	
	B	25	350	1000	12°		.0781	L	8400	
10		39	300	925	$10\frac{1}{2}°$		.1245	L-C	11050	

DIMENSIONS OF CONICAL CLUTCHES

IDENTIFICATION NUMBER		DIMENSIONS OF CONE: ANGLE OF FACE	DIAMETERS: OUTSIDE	DIAMETERS: INSIDE	DIAMETERS: MEAN	WIDTH OF FACE	AREA IN CONTACT	FRICTION MATERIAL: CONE	FRICTION MATERIAL: FLYWHEEL	SPRING: OUTSIDE DIA.	SPRING: THICKNESS	SPRING: LENGTH	SPRING: COMPRESS	SPRING: NUMBER
1		12½°	15¼	14¼	14¾	2½	116	L	CI	3⅛	⅜	3¼	200	1
2		13°	14½	13½	14	2¼	99	L	CI	2½	5/16	2¾	350 to 400	1
3		10°	14⅝	13⅝	14⅛	3⅛	138.6	L	CI	1 3/16	⅛	2 7/16	93 to 100	6
4	A	12°	16.395	15.395	15.895	2 7/16	121.7	L-C	CI	1⅞	¼	2	640	1
	B	12°	18.00	17.00	17.50	2 9/16	141.0	L-C	CI	1⅞	¼	1⅞	640	1
	C	12°	20⅝	19.½	20 1/16	2 15/16	184.6	L-C	CI	1⅞	¼	2	640	1
5		10°	15¼	14¼	14¾	3¼	150.1	L	CI	2 5/16	¼	4 7/16	300	1
6	A	10°	18⅜	17¼	17 13/16	3¼	182.0	L	CI	3¼	5/16	3⅞	450	1
	B	10°	16¼	15⅛	15⅝	3¾	184.8	L	CI	3	5/16	3	225	1
7	A	11°	14⅝	14	14 5/16	1 15/16	87.25	L	CI	1 5/16	5/16	2¾	350	1
	B	11°	19	18⅛	18 9/16	2 3/16	127.50	L	CI	2 13/16	11/32	2⅞	400	1
8		10°	19½	18⅝	19 1/16	2½	149.4	L	CI	1 9/16	5/32	10⅞	350	1
9	A	12½°	15¾	14½	15⅛	2⅝	124.70	S	CI	2 11/16	¼	2⅞	400	1
	B	12°	13	12	12½	2½	98.25	S	CI	2½	7/32	3¾	350	1
10		10½°	16½	15⅝	16 1/16	2 5/16	116.3	L-C	CI	4⅛	⅜	3½	300	1

CHAPTER III

Disc Clutches.

8. Single Disc. - Figure 2 shows a sketch of a single disc or plate clutch as it is called.

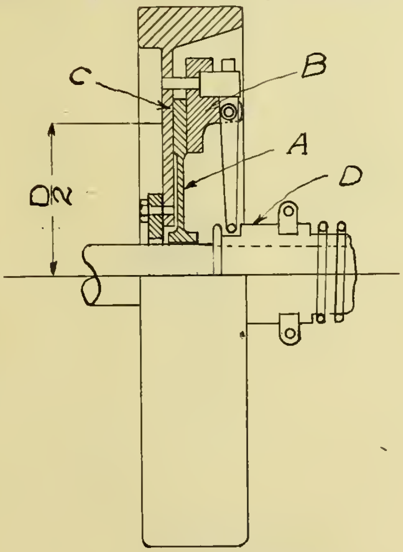


FIGURE 2.

The plate A is keyed to the transmission shaft, and is finished on both faces. Part B represents the presser which is attached to the fly wheel by means of a bolt as shown. Attached to B is a lever arm which engages with the sliding sleeve D. By moving D axially in either direction, the pressure may be applied or released on the plate A. A spring as shown holds the clutch in engagement. Between the plate A and the flywheel C and the presser B there is generally some friction material such as raybestos or fibroid which gives a high coefficient of friction, and at the same time a better wearing surface. Plate III is a

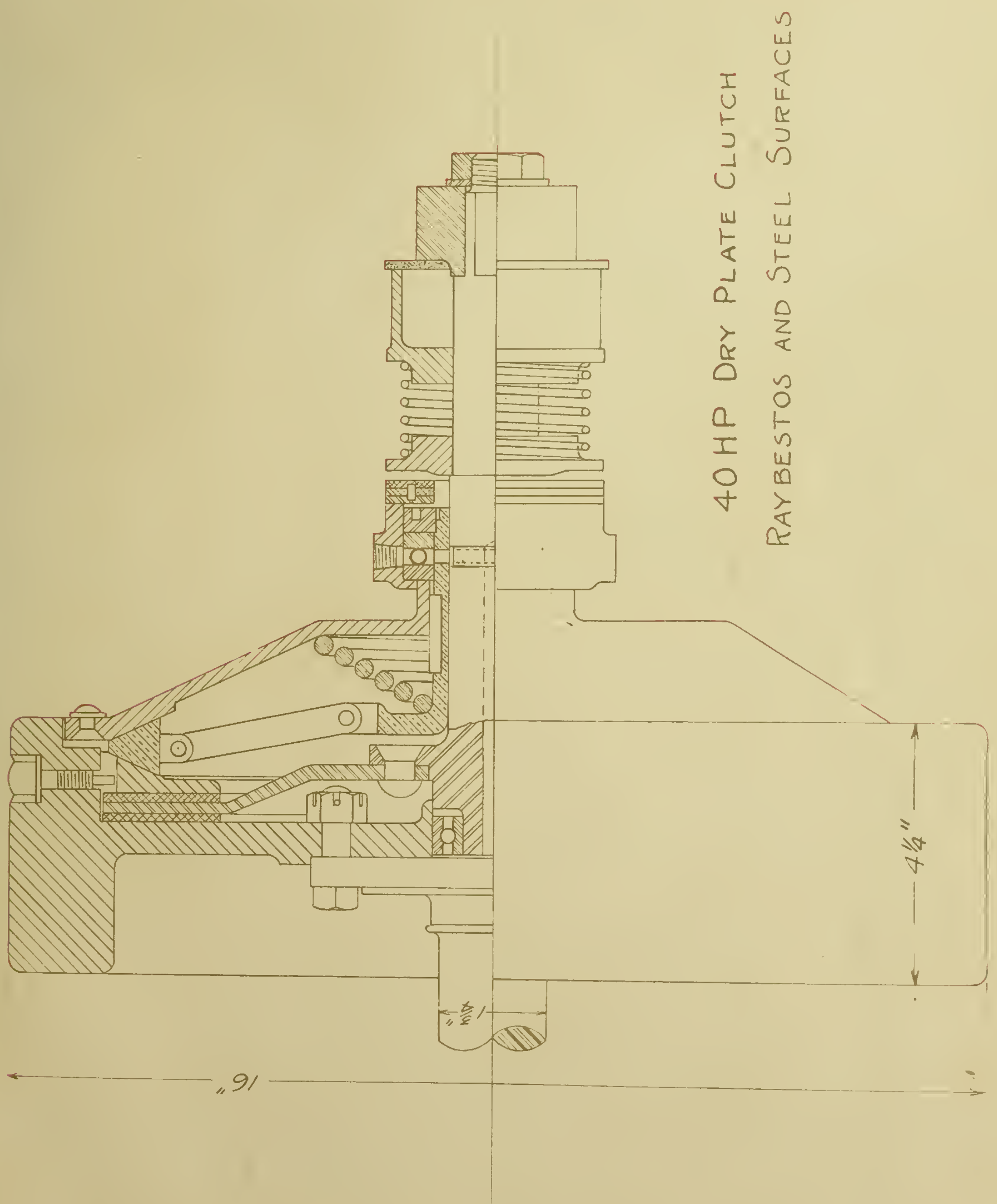


PLATE III

reproduction from a drawing of an actual plate clutch used in a first class car.

In this clutch the plates are pressed together by a wedge which is connected by a toggle joint to the sliding sleeve on the shaft. The driven plate is steel and the friction surfaces are raybestos. Adjustment can be made by means of the set screw which raises or lowers the wedge. The advantage of the plate clutch lies in the fact that having less weight, the inertia force is less than that in a cone clutch. There being only two friction surfaces there is not the tendency to drag which is common in the ordinary multiple disc clutch. The spring in Plate III is on the shaft and is of the conical spiral type. Other makers use a helical spring on the shaft, and some use three springs placed on bolts or studs on the flywheel.

To determine the relation that exists between the torsional moment transmitted, the dimensions of the discs, the axial load applied, and the coefficient of friction, we shall assume that the normal wear of the discs is proportional to the work of friction.

Let D = the mean diameter of discs.

P = axial force exerted.

According to the above assumption the moment of friction for a single surface in contact is given by the following formula

$$M = \frac{\mu P D}{2} \qquad (4)$$

To calculate the moment when n surfaces are in contact simply multiply equation (4) by n . Changing the form of (4) by solving for μ we have for n surfaces in contact

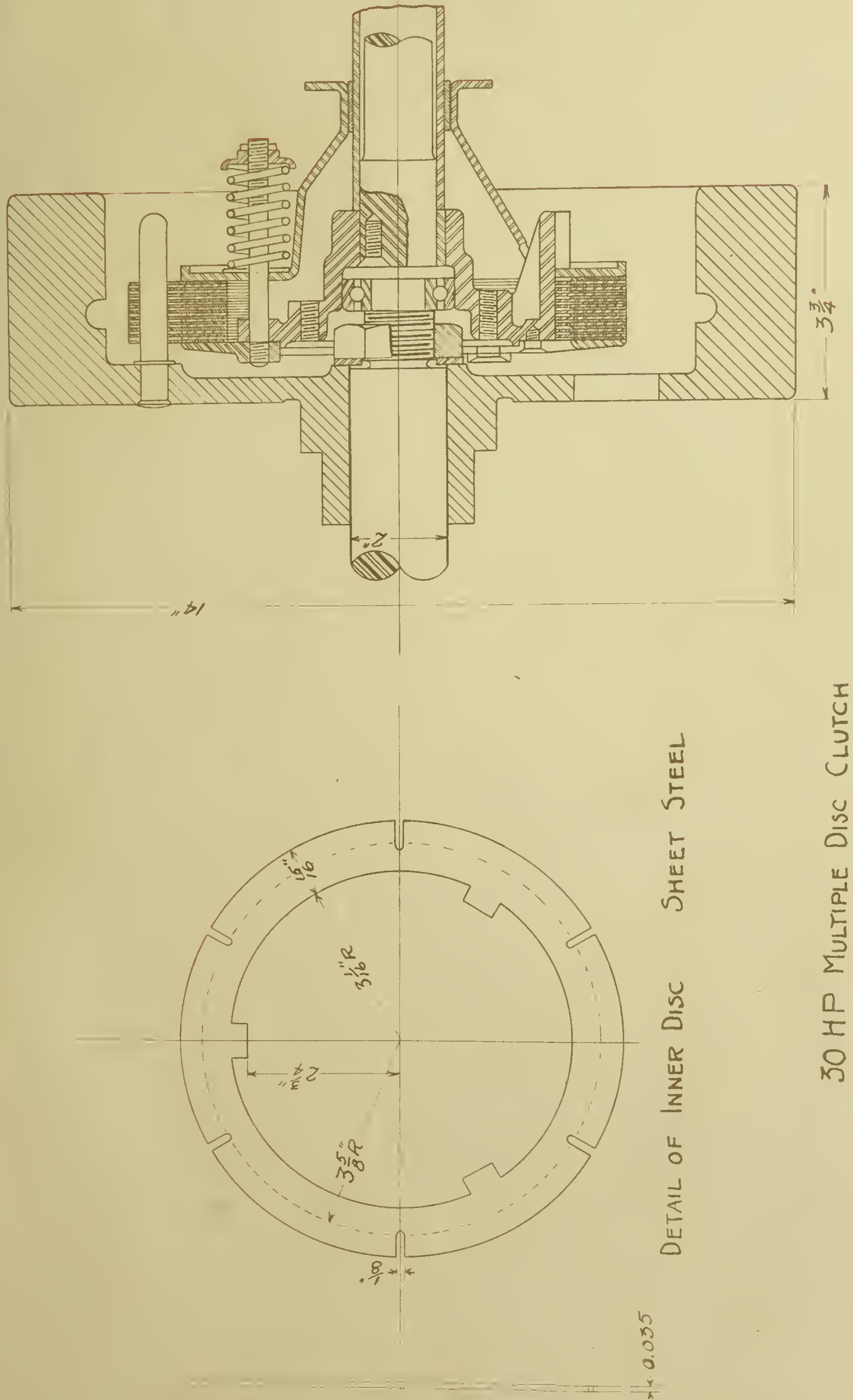

14"
2"
3¾"
5/16"
3 1/16" R
2¾"
3 5/8" R
1/8"
0.035
DETAIL OF INNER DISC   SHEET STEEL
30 HP MULTIPLE DISC CLUTCH

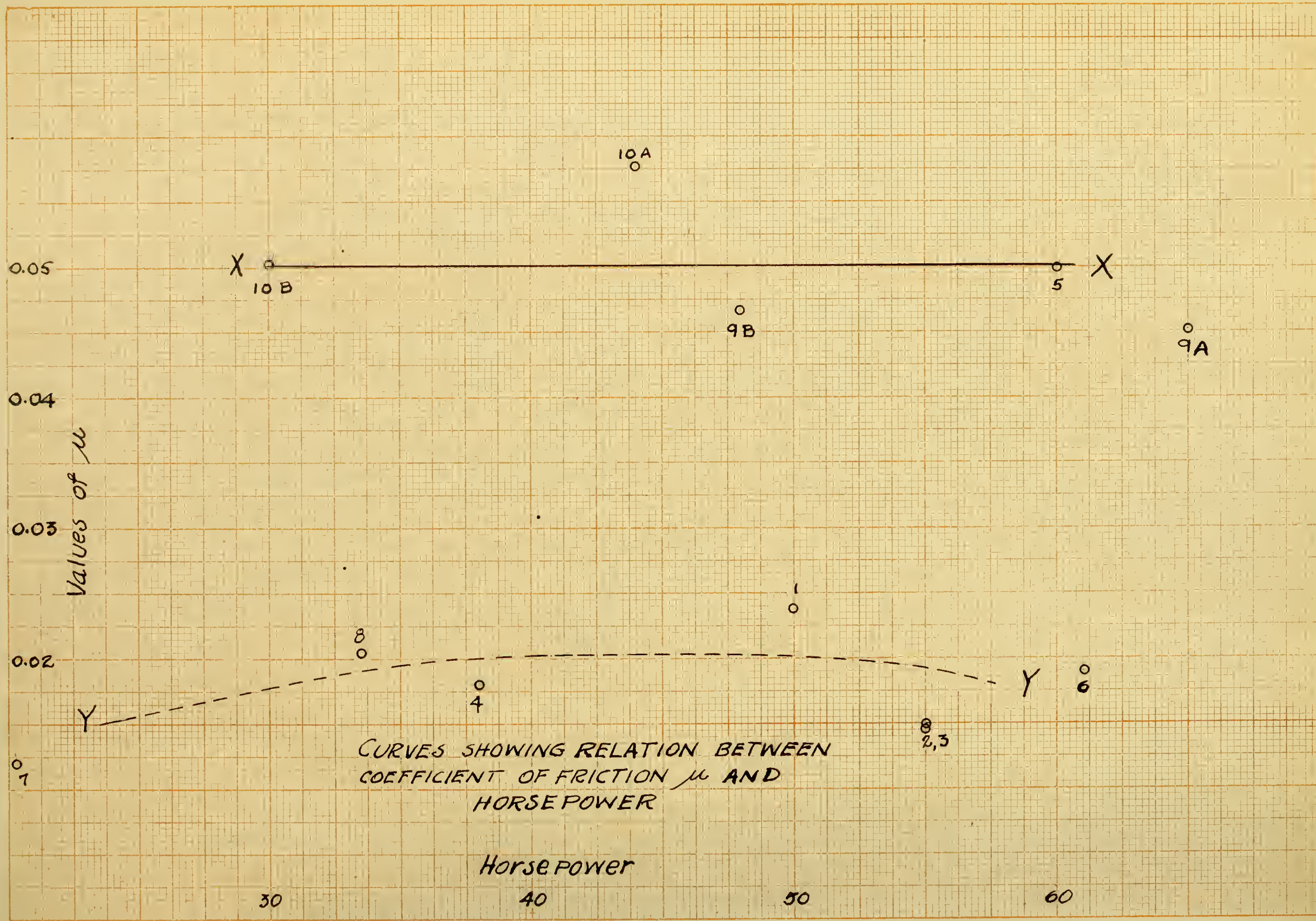
PLATE IV.

$$\mu = \frac{2\,M}{D\,P\,n} \qquad (5)$$

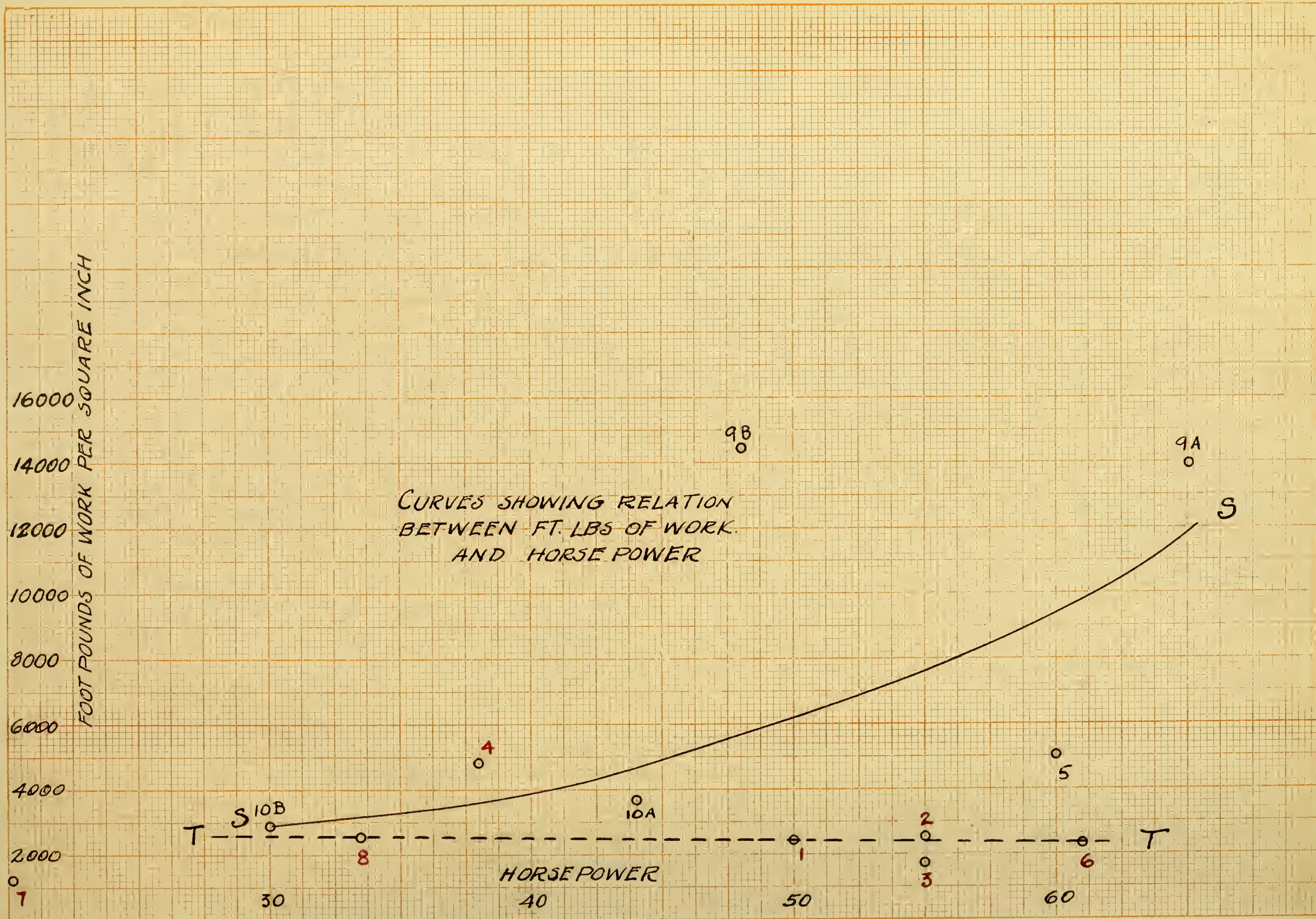
The number of footpounds of energy absorbed per square inch of contact surface is

$$W = \frac{33{,}000 \text{ H. P.}}{\text{area}} \qquad (6)$$

9. Multiple Disc Clutch. - Plate IV. shows a multiple disc clutch with steel discs. The driving or outer discs are mounted on a stud in the flywheel as shown in the figure. The driven discs are fastened to the sliding sleeve by the projections shown in the detail of the inner disc. The plates are held in contact by three helical springs placed 120° apart. Between the plates and the spring is a presser plate. This is connected to a sleeve which is actuated by the pedal. A ball bearing helps to hold the plates in alignment. The notches shown in the flywheel rim are holes bored in on a slant, and whose purpose is to aid in ventilating the clutch and thus cool the discs. A majority of the manufacturers use some sort of device to separate the plates when the clutch is thrown out. The most common form is to bend small projections on the plates. These form small springs which push the plates apart when the spring pressure is relieved. The radius of the discs should be as small as possible in order to cut down the inertia force. The largest diameter received was 13 3/4 inches and the smallest was 7 1/8 inches. The chief objection to the disc clutch is the tendency to drag after the clutch is thrown out. This is caused by the great number of discs which are in such close contact that some little time elapses before all are separated. To eliminate this dragging action some manufacturers use a clutch brake. A



CURVES SHOWING RELATION BETWEEN COEFFICIENT OF FRICTION μ AND HORSEPOWER
Values of μ
0.05
0.04
0.03
0.02
Horsepower
30
40
50
60
X
X
Y
Y
10A
10B
9B
9A
5
1
8
4
2,3
6
7


CURVES SHOWING RELATION
BETWEEN FT. LBS OF WORK
AND HORSE POWER
FOOT POUNDS OF WORK PER SQUARE INCH
HORSE POWER
16000
14000
12000
10000
8000
6000
4000
2000
30
40
50
60
S
T
10B
10A
9A
9B
1
2
3
4
5
6
7
8

great many disc clutches run in oil which spreads itself in a thin film over the surfaces of the plates and is squeezed out when the clutch is engaged. Frequently some sort of friction material is used between the plates such as raybestos, fibroid, etc.

10. Discussion and Conclusions. - On page 21 are two curves showing the relation between the coefficient of friction and the horsepower. Curve X X may be assumed to represent an average curve for disc clutches with raybestos facing between the plates. The tendency seems to be to keep the coefficient of friction constant for all horsepowers and increase the work per square inch as shown by a curve S S on page 22 . The coefficient μ is very much higher for clutches using the friction material between plates. Y Y shows values of μ for the plates without the special facing and this also seems to approximate a horizontal line between 40 and 50 horsepower, but it seems to drop off on either side of this. Curve T T on page 22 represents an average curve for the work done per square inch of friction area and it shows a tendency to slope downward with an increase in the horse power transmitted. This probably means that at the higher horsepower, the designers prefer to provide more area and a lower coefficient of friction.

On page 25 are tables of disc and plate clutches giving the principal dimensions and important data.

The plate clutches have a higher value for the energy transmitted per square inch of area. Clutch No. 11 in the table on pages 25 and 26 develops 19,100 foot pounds of work per square inch and No. 12 develops 20,600 foot pounds per square

inch. These high values can be used since there is a facing of raybestos or some other friction material between the driving and driven plates. Values for the coefficient of friction lie within the limits calculated for ordinary disc clutches.

Dimensions of Disc and Plate Clutches

Identification No		Dimensions of Plates: Outside Dia	Inside Dia	Mean Dia	Thickness	Friction Area	No of Plates: Driver	Driven	Material: Driver	Driven	Spring: Outside Dia	Thickness	Tension	Number
1		10"	9	9 1/2	0.044	686.5	25	26	SAW ST	SAW ST	2 1/2	5/16	220	1
2		7 1/8	6 1/8	6 5/8	1/32	687.	34	33	s	s	1 1/16	5/32	250	3
3		7 3/4	6	6 7/8	0.04	1060	28	29	s	s	2 1/4	1/4	350	1
4		7 3/4	6 7/8	7 5/16	0.065	261.5	14	13	s	s	7	1/4 × 1/2	250	1
5		10	8	9	1/8	395.5	7	8	SAW ST	SAW ST	6	5/16	450	1
6		11 1/2	10	10 3/4	.049	862.5	18	17	s	s	4 3/16	5/16	450	1
7		8 3/32	6	7 3/64	1/16	532.5	12	12	s	s	3 1/8	1/4	242	1
8		8	6 1/8	7 1/16	.03196	417.	11	10	PH BR	SH'T ST	1 1/8	1/8	270	3
9	A	8	6	7		154	7	8	s	s			450	1
	B	8	6	7		110	5	6	s	s			400	1
10	A	13 3/4	11 1/4	12 1/2	0.08	393.5	5	4	s	s	1	1/8	500	3
	B	13 3/4	11 1/4	12 1/2	0.075	344	4	4	s	s	1	1/8	500	3
11		14 7/8	9 1/2	12.1875	5/16	103.8	2	1		CORK INS	1	1/8	350	4
12.		12 3/4	9	10 7/8	3/16	64.2	2	1		s	3	5/16	180	1

Table of Results

Identification No.		Horse-power	RPM	Spring Tension	Friction Area	Material	Value of μ	Ft. Lbs of Work per Sq Inch: Single	Total	Formula	Remarks
1		50	1000	220	686.5	S	.0239	50	2400	$\mu = \frac{2M}{DPn}$	
2		55	1200	250	687.0	S	.01462	40	2540		
3		55	1000	350	1060	S	.01475	30.6	1714		
4		38	1700	250	261.5	S	.01805	184.6	4800		
5		60	1000	450	395.5	S	.05	358.3	5020		RAYBESTOS
6		61	1100	450	862.5	S	.01905	68.5	2330		
7		20	1400	242	532.5	S	.01292	54.0	1240		
8		33½	1600	270	417.	PHOS. BR.	.0204	132.5	2650		
9	A	65	1200	450	154	S	.0452	993.	13900		RAYBESTOS
	B	48	1200	400	110	S	.04675	1440	14400		"
10	A	44	1000	500	393.5	S	.0578	461	3690		"
	B	30	1000	450	344	S	.0502	410.	2875		"
11		60	1200	220	103.8	S	.1125	9550	19100		"
12		40	1000	180	64.2	S	.1310	10300	20600		"

CHAPTER IV

Combined Disc and Cone Clutch.

11. Hele-Shaw Clutch. - Plate V page 28 shows a Hele-Shaw clutch. As can be seen in the drawing it is in the plates that the difference between the ordinary multiple disc clutch and the Hele-Shaw clutch lies. The plates are of steel but have a double cone shaped projection pressed or stamped into them. These cones have a contact area approximately 3/8 of an inch wide, and it is claimed that this the limit for proper lubrication. The plates are entirely encased and run in oil. Fig. 3 shows the forces acting on the conical projections.

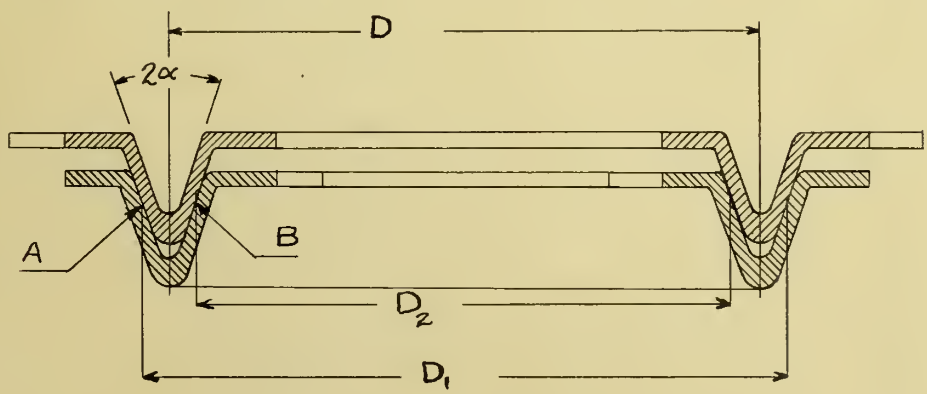


FIGURE 3.

12. Force Analysis. - The following formulas are taken from Mechanics of Machinery Part I. The moment of friction at A is

$$M_1 = \frac{\mu P D_1}{4 \sin\alpha}$$

The moment of friction at B is

$$M_2 = \frac{\mu P D_2}{4 \sin\alpha}$$

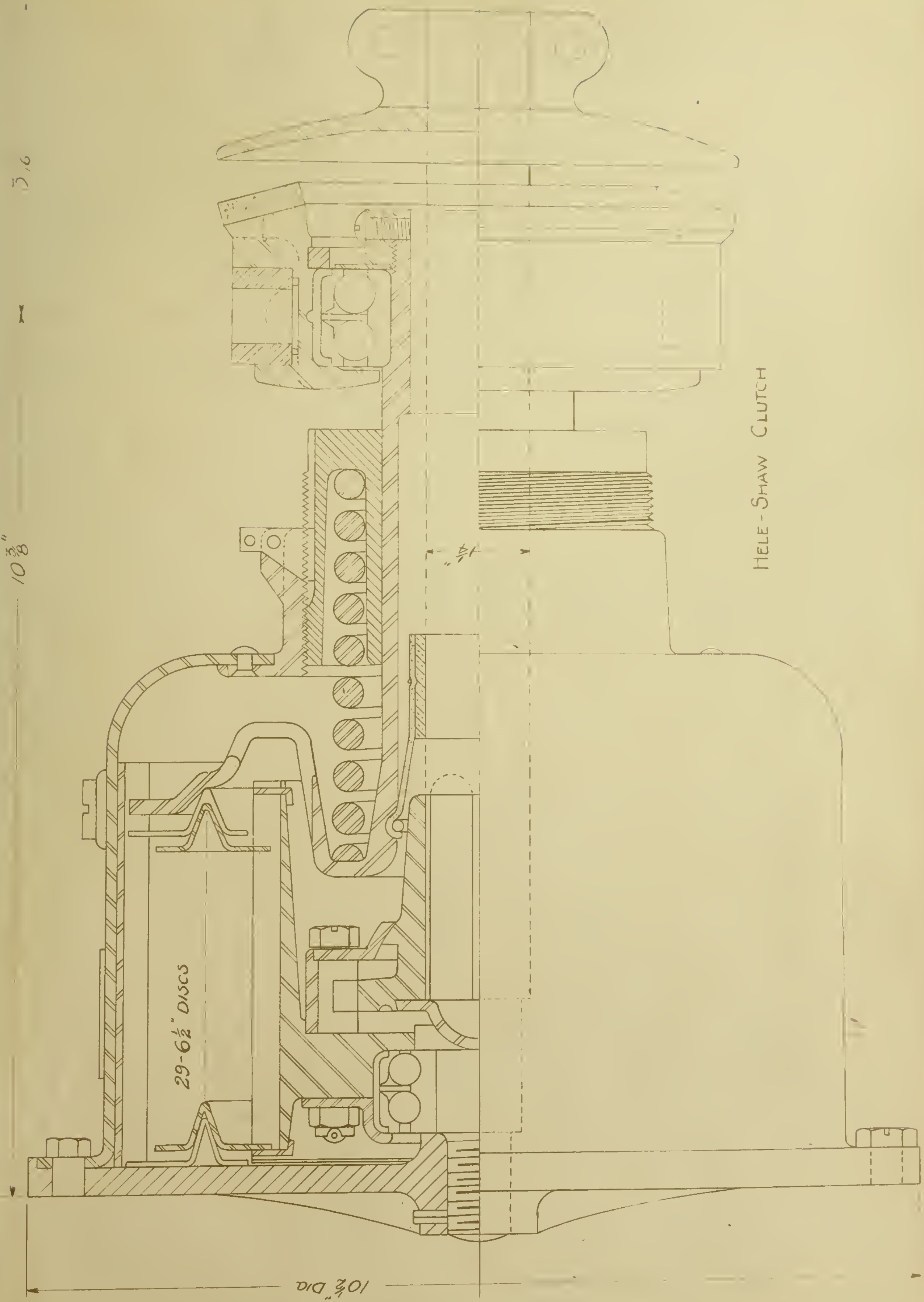


PLATE V.

The total moment of friction for one contact surface is the sum of M_1 and M_2 and for n surfaces

$$M = \frac{\mu P\ n}{4\ \sin\alpha}\ (D_1 + D_2) \qquad (7)$$

Representing $\frac{(D_1 + D_2)}{2}$ by D, (7) reduces

to $$M = \frac{\mu P\ D\ n}{2\ \sin\alpha} \qquad (8)$$

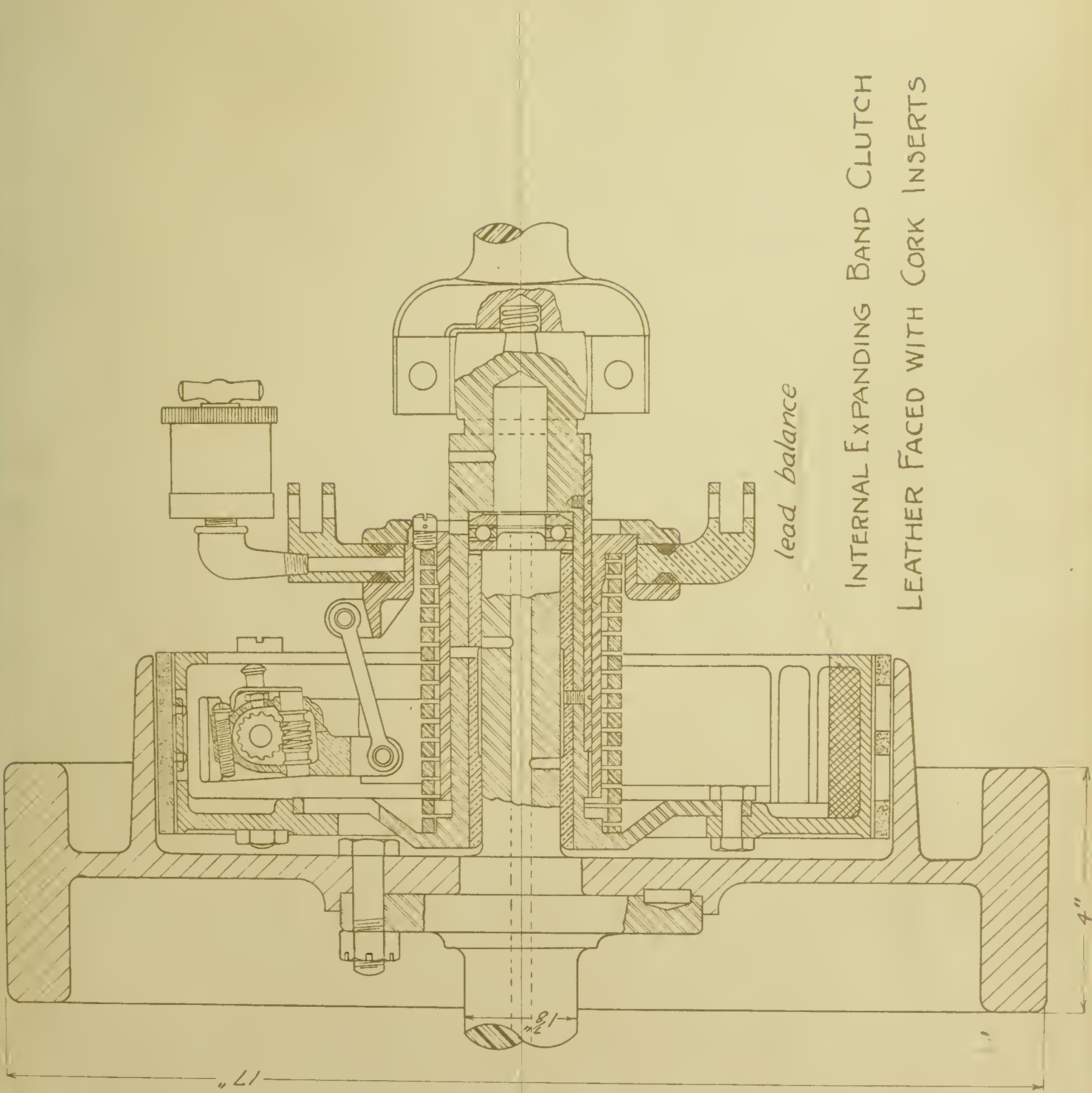
Solving for the coefficient of friction we have

$$\mu = \frac{2\ M\ \sin\alpha}{n\ D\ p} \qquad (9)$$

13. Results. - Below are given dimensions of the Hele-Shaw clutches examined.

Size	H.P. at	No Plates	Mean Dia.	Friction Area	Angle Cones	Spring Press
5	30	29	$6\frac{1}{2}$	430	35°	175 to
6	40	23	$8\frac{1}{2}$	440	35°	250

For the 30 H.P. clutch the value for μ is 0.00825 and the work transmitted per square inch of friction area is 2300 foot pounds. For the 40 H.P. clutch μ is 0.01432 and the work per square inch is 3300 foot pounds. The required spring pressure is lower than in the ordinary disc clutch due to the conical surfaces. The plates being separated by an air space at all times there is better dissipation of heat than in the plate or disc clutches where the plates are all packed together solidly.



Internal Expanding Band Clutch
Leather Faced with Cork Inserts
lead balance
4"
17"

PLATE VI.

CHAPTER V

Rim Clutches

14. Internal Expanding Band. - Below is shown a sketch of an internal expanding band clutch with the forces acting upon it.

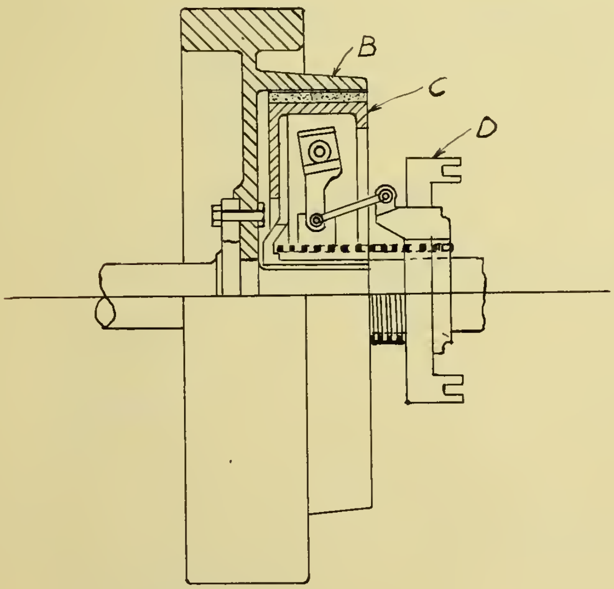


FIGURE 4.

The drum B is cast integral with the flywheel web and is finished on the inside. The band C is split and can be expanded by means of the screw and pinion shown. By moving the part D axially, the band C is either contracted or expanded as the case may be. The spring mounted on the shaft holds the clutch in engagement. As in the cone clutch the facing of the rim clutches may be either leather or leather with cork inserts. The clutch shown uses cork inserts set in the leather facing. The drawing from which this sketch was made is shown in reduced size on page 30 and

is taken from a clutch in use on one of the high priced cars. In the drawing it will be noticed that the expanding mechanism is counterbalanced with lead.

In developing the formula for this clutch it is assumed that the pressure is uniformly distributed over the entire surface in contact. This may not be correct when the clutch is first thrown in, but after a few turns when no more slipping occurs it may be assumed that practically the entire surface is in contact and that the pressure is uniformly distributed. According to our Mechanics of Machinery Part I the moment of the friction force is

$$M = \frac{\mu p_1 \, l \, d^2 \pi}{2} \qquad (10)$$

where
p_1 = presure per sq. inch of projected area.
l = length of face.
d = diameter of drum.
μ = coefficient of friction.

The horsepower transmitted at N revolutions per minute is

$$H = \frac{\mu \pi^2 p_1 \, l \, d^2 N}{396000} \qquad (11)$$

Since μ and p_1 are constant for any case, their product may be represented by some new constant K

Hence $$H = \frac{K l d^2 N}{40120}$$

and $$K = \frac{40120 \, H}{l \, d^2 N} \qquad (12)$$

Below are given the principal dimensions and data on the two expanding band clutches received.

	H. P.	R. P. M.	l	d	K
A.	70	1600	3	12	48.6
B.	60	900	2.35	15.28	68.3

Rim clutches in factory use, have values of K from 50 to 100. Clutch A has cork inserts while B uses plain leather facing. The users of the expanding band clutch claim that there is an advantage in this type over the external contracting type in that the centrifugal force developed tends to expand the band still further and prevents any tendency to slip. It is doubtful whether this is true inasmuch as the engine torque is at all times the driving force and the greatest frictional force is developed at low speed when the centrifugal force is small.